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ABSTRACT

In a refrigeration screw compressors, bearing lubrication refers to the elasto-hydrodynamic, (EHL), regime. Under these conditions, bearing life is dependent on the film thickness separating the surfaces and the stress produced by the cyclic load on the surface material. Since the bearings in a refrigeration screw compressor are lubricated with a mixture of compressor oil diluted with the refrigerant, it is important to understand the behaviour of such mixtures. It is known that the presence of refrigerant decreases the viscosity and pressure-viscosity coefficient of the lubricant; those two lubricant properties which are most important in ensuring a film thickness sufficient to separate the contact surfaces in an EHL contact. The parameter controlling the location of the maximum stress in the material and, to some extent, the magnitude of that stress, is the friction coefficient in the contact. In this investigation the influence of refrigerant dilution on friction coefficient in a contact at a pressure of 2.5 GPa has been studied.

The investigation showed that presence of refrigerant increase the friction coefficient in the lubricant.

INTRODUCTION

In a refrigeration compressor, the bearings supporting the rotors are highly loaded and lubrication is in the EHL (elasto-hydrodynamic-lubrication) regime. Under such lubricating conditions, a very thin lubricant film and the materials in the bearing surfaces are exposed to a high contact pressure that results in the material deforming elastically in the contact. The lubricant in an EHL contact is also subjected to high shear rates and, due to the high local pressures, lubricant compression can occur.

In a refrigeration screw compressor the lubricating medium is oil diluted with the refrigerant used in the refrigeration system. It has been shown by Tuomas and Jonsson[1] that the presence of refrigerant lowers both the viscosity, η, and pressure-viscosity coefficient, α, of the lubricant and hence the film thickness in the contact. Also in refrigeration applications, Jonsson[2] showed that the use of esters with different molecular branching influence η and α differently. A more branched molecule results in an increases in η and α and therefore, according to Hamrock and Dawson[3,4], will result in a thicker lubricating film.

Figure 1 shows the inner ring of an angular contact bearing (7210). The bearing failed due to surface fatigue after 1900h. The lubricant used in the experiment was a polyolester oil diluted with 26% R-134a. The temperature was 40°C, speed 1500rpm and load ratio C/P 12.4.

The fatigue life of a surface depends on the number of load cycles the surface material is exposed to, Lundberg and Palmgren[5]. If the stress intensity becomes too high, cracks will start to develop from dislocations or other inhomogeneous parts of the material. An important determining factor of the fatigue life is the depth of the maximum stress. If the maximum is located close to the surface, the material will fail more easily than if the maximum is located deeper in the material. Sliding in the contact will add another stress component to the combined stress that act upon the material, τ_L. τ_L, showed by Jacobson and Hamrock[6] to be the limiting shear stress that the oil can sustain. The friction coefficient of the solidified lubricant, τ_L, combined with the pressure, P, is called the shear strength-pressure coefficient [γ] (friction coefficient of the solidified lubricant) and is given by [$\gamma = \frac{\partial \tau_L}{\partial P}$]. It was shown by Höglund[7], that higher γ causes the stress maximum to move closer to the surface of the material and also increase the maximum stress slightly.

This paper investigates how shear strength-pressure coefficient [γ] is affected by refrigerant concentration in the oil.

METHOD

To be able to measured the friction coefficient of the lubricant under pressures up to 3 GPa an apparatus described by Åhrström[8,9] was used, the apparatus show in Figure 2, considers an oblique impact of a steel ball on a rod end. Accelerometers and strain gages measure the shock waves and from that information the transient normal- and transverse forces, acting during the impact, are calculated. A change in friction coefficient affects the wave pattern and consequently also the forces on the end plane of the rod.

Test procedure

The ball and the rod end surface were thoroughly cleaned and a lubricant sample placed on the tungsten carbide plate located on the end of the rod. The ball was then released in such a way that an oblique contact resulted, the height of release being such that the required contact pressure of 2.5 GPa was achieved. In the experiments 2.5 GPa was used to get full film separation in the contact, a lower contact pressure makes the lubricant to collapse and metal-to-metal contact occurs.

To be able to detect if it is a full film separation between the ball and the rod end, a potential difference of 100 mV is applied between the ball and the end plane. Any potential equalization, resulting from asperity contact, is monitored on an oscilloscope. Boundary lubrication contact measurements are not relevant because metal-to-metal contact occurs at the impact, and are therefore excluded.

Tested lubricants and refrigerants

Three different polyolesters lubricating oils were used; Ester A, B and C. All three are polyolester refrigeration oils but with different viscosity grades and acid branching. The ester oils were diluted with the HFC refrigerant R-134a. A mineral oil diluted with the chlorinated refrigerant R-22 was used as a reference. Details of all these oils are given in Table 1 together with the refrigerants used. The molecular structures of the esters are shown in Figure 3.

Refrigerant concentration estimation

At ambient pressure, the refrigerant will boil out of the oil. In order to know the actual concentration at the time of the friction measurement, experiments were carried out to determine the change in refrigerant concentration at ambient pressure over time for the four different mixtures. 27 grams of oil were placed in a small bottle and then diluted with 5 grams of refrigerant. The pressure in the bottle was then decreased until the refrigerant started to boil out of the oil. The change in weight was then measured over time and the concentration calculated. The results are shown in Figure 4. As can be seen, refrigerant concentration at zero time varies with the molecular structure of the oil; a more branched acid in the ester results in higher solubility. Henderson[10] also describes this.

Once the sample is placed on the tungsten carbide plate the refrigerant will begin to evaporate. Since the time of the measurement in the oblique impact test rig was known, the refrigerant concentration in the oil at the time of measurement could be calculated for each oil mixture from Figure 4. These concentrations are given in

Table 2.

RESULTS AND DISCUSSION

In Figure 4 the change in refrigerant concentration over time for the tested lubricants is shown. It can be seen that the amount of R-22 diluted in the mineral oil decreased faster than the R-134a in ester. This means that the mineral oil will more easily give up any diluted refrigerant if the pressure is lowered and therefore provide an increased film thickness in the bearings.

Figure 5 shows the friction coefficient of the tested oils both undiluted and diluted with refrigerant. Each box represents five measurements and the height of the box indicates the measurement deviation. This deviation is due to refrigerant boiling out of the oil during the experiment. The experiments were carried out at ambient pressure at a temperature of 21°C with a contact pressure of 2.5 GPa. The actual refrigerant concentrations were between 4.5% and 8.5%. See

Table 2.

The results shown in Figure 5 clearly show that the coefficient of friction in the contact increases when refrigerant is present. The molecule structure of different esters also affects the friction in the contact. Long straight molecules deform more easily when subjected to shear stress that results in a lower coefficient of friction in the contact. Ester C, which contains 100% branched acids, had the highest friction coefficient of all the tested lubricants.

The lubricant acting in the contact can be in three forms; liquid-, visco elastic- and solid phases, Åhrström et al[11]. The measured friction coefficients are thus a combination of the effects of the three phases, namely, the solidified friction coefficient [γ], the visco elastic friction coefficient and the friction coefficient of the fluid. The dominant component for the oil/refrigerant mixtures in these experiments is from γ. Adding refrigerant to the oil causes the viscosity to decrease, which will also reduce the friction coefficient component from the liquid phase. The effect of solidified friction coefficient γ, which is far higher than that of the liquid, thus increases.

The experiments were carried out at ambient pressure and temperature and can therefore only give an indication of the influence the refrigerant on the friction coefficient. Water from the atmosphere also dilutes the ester oils.

CONCLUSIONS

Measurements of the strength-pressure coefficient [γ] at extreme pressures have been made on four refrigeration oils; either in there undiluted form or diluted with refrigerant. The objective of the work was to see whether γ changes when the oils were diluted with refrigerant. Since γ is coupled to the limiting shear stress, any increase in γ will also increase the maximum stress in the material and also cause the maximum stress to act closer to the surface.

The results show that, at a pressure of 2.5Gpa, γ increases with increasing refrigerant concentration at least up to a level of ca. 9%. Since in actual refrigeration conditions refrigerant concentrations in the lubricating oil can be up to 30%, experiments at these levels of dilution must be carried out.

The investigation also agrees with the results of earlier investigations which indicated that molecule structure was an important parameter for achieving the correct strength-pressure coefficient [γ] for a given application.

The work also showed that the concentration of R-22 in mineral oil decreased much faster than the R-134a in the ester mixtures.

ACKNOWLEDGEMENTS

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FIGURES

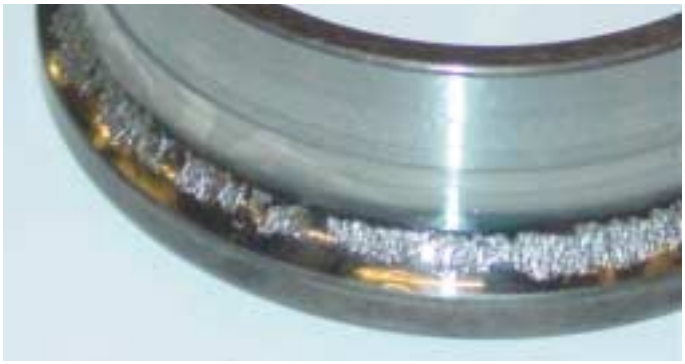

Figure 1 An angular contact bearing (7210) that failed after 1900h running with a polyolester oil diluted with R-134a at a concentration of 26%. Operating parameters, 1500rpm, temperature 40°C and load ratio 12.4.

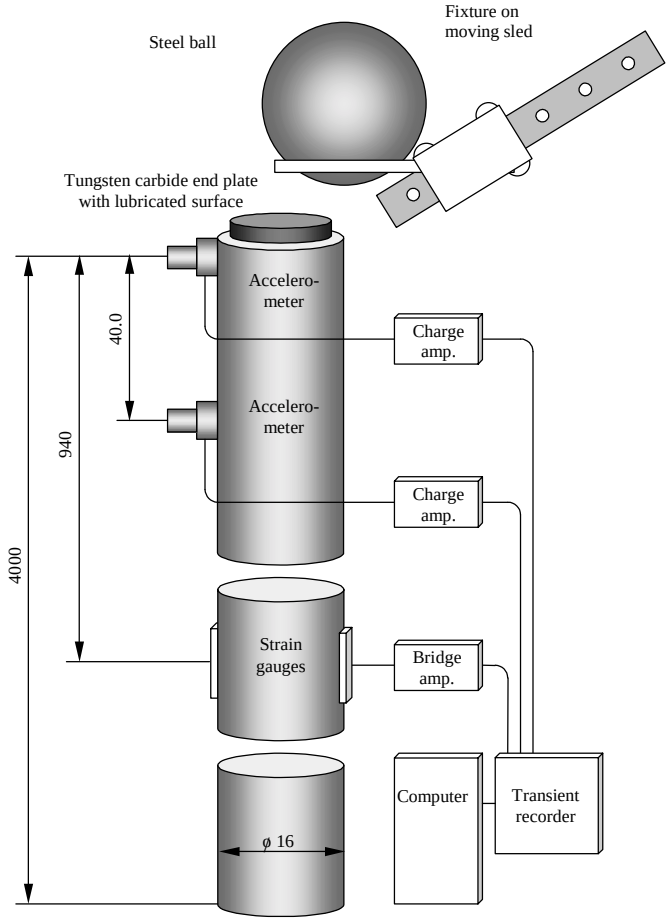

Figure 2 The test apparatus (Åhrström[8,9]) used to measure lubricant friction coefficients at pressures of up to 3GPa.

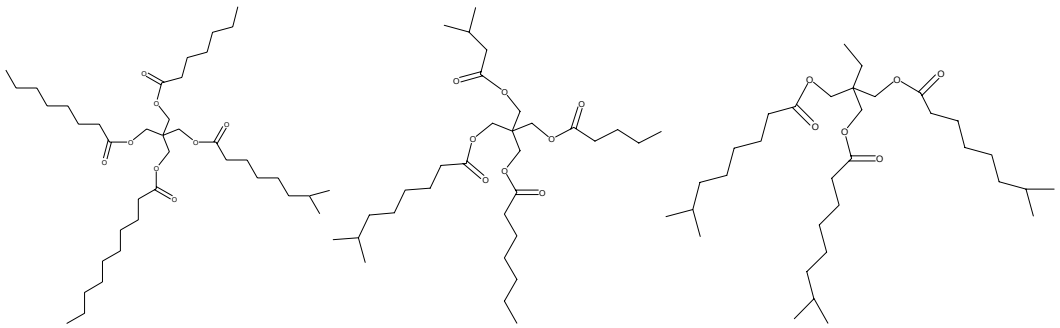

Figure 3 Molecular structure of the tested polyolesters. From left to right, Ester A, B and C.

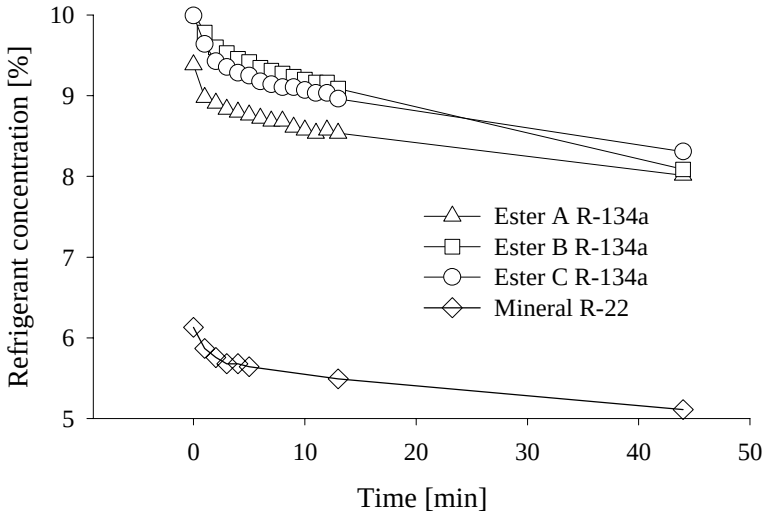

Figure 4 Refrigerant concentration at ambient pressure vs. time for the oil/refrigerant mixtures tested.

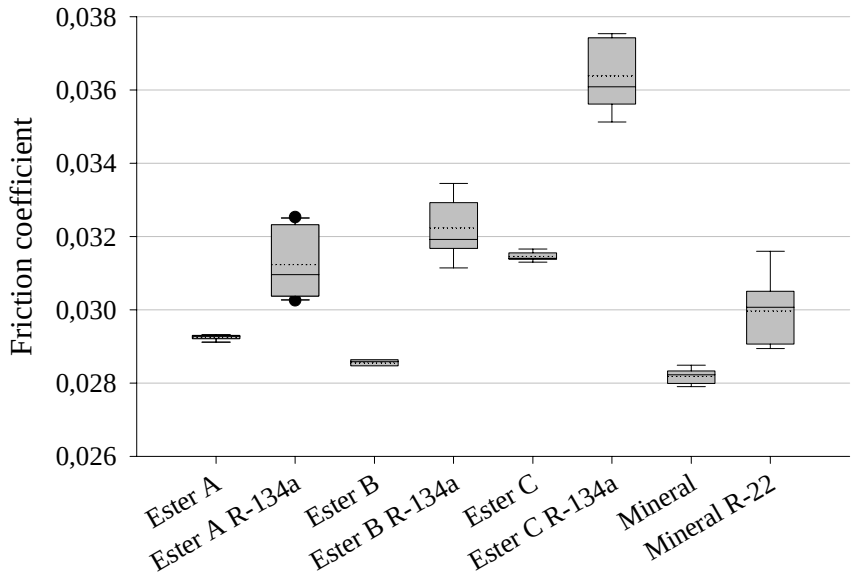

Figure 5 The friction coefficients of the undiluted oils and oils diluted with refrigerant.

TABLES

Table 1 Details of the four lubricants used in the experiments.

Lubricant	Chemical structure	Viscosity grade	Branched acids	Refrigerant used
Ester A	$C_{39}H_{72}O_8$	VG68	70%	R-134a
Ester B	$C_{31}H_{56}O_8$	VG68	8%+67%	R-134a
Ester C	$C_{33}H_{62}O_6$	VG46	100%	R-134a
Mineral	-	Base viscosity 57cst at 40°C	-	R-22

Table 2 Refrigerant concentrations used in the experiments.

Oil	Refrigerant	Refrigerant concentration [wt.%]
Ester A	R-134a	7 %-8 %
Ester B	R-134a	7.5 %-8.5 %
Ester C	R-134a	7.5 %-8.5 %
Mineral	R-22	4.5 %-5 %